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THEORETICAL CONSIDERATIONS REGARDING THE DYNAMIC ABSORBER

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Abstract: *The paper is a theoretical one that focuses on the influence of dynamic absorbers in the reduction and elimination of torsion vibrations. The dynamic absorber is used for the reduction of the amplitude of the torsion vibrations of crankshafts. According to the construction restrictions, we notice that the dynamic absorber does not increase the number of resonance phenomena, irrespective of the order of the harmonic that causes forced vibrations.*

Key words: *dynamic absorber, torsion vibrations, reduced masses.*

1. INTRODUCTION

The dynamic absorber is an oscillation system that is attached to a system with the purpose of reducing its vibrations. The crankshaft of a piston-based engine, together with its gearings, forms an oscillating system. The oscillation amplitudes can become very dangerous when they resonate with the device, which is when the rotation of the shaft becomes the same with one of its critical rotations.

By adding a dynamic absorber to a device, the primary system's degrees of freedom increase by a unit and thus the resonance curve is modified. If the parameters of the dynamic absorber are properly chosen, the resonance curve of the system composed of primary system and absorber has a minimum declared value in a frequency area in which the primary system has only one maximum. Thus, there is a significant damping in the primary system vibrations in the vicinity of its resonance.

The dynamic absorber can be designed at the same time with the structure whose vibrations must be eliminated or can be added later on. Also, it can act upon the entire structure or can be attached to the element whose functioning produces vibrations in the overall structure.

The dynamic absorber is also used to reduce the amplitude of the torsion vibrations in the case of crankshafts. In order to calculate the torsion vibrations of such a complex elastic system, this system has to be turned beforehand into a simpler equivalent dynamic system, formed of an elastic linear shaft of negligible mass loaded with circular reduced masses obtained by the reduction of the mobile gears.

This paper is a theoretical one that presents the study of mechanical systems of five reduced masses that receive a dynamic absorber placed at the and of the mechanical system. The harmonic x acts upon the mechanic system is applied to the reduced mass m_1, m_2, m_3, m_4 and m_v.

The mass that receives the dynamic absorber is acted upon with an order x harmonica which results by the decomposition of the Fourier series of the disruptive force with a periodical variation acting upon this mass.

2.THE MECHANICAL SYSTEM FORMED OF FIVE REDUCED MASSES, THE DYNAMIC ABSORBER PLACED AT THE AND OF THE SYSTEM

The case takes into account is a mechanical system formed of five reduced masses system noted m_1, m_2, m_3, m_4, m_v (Fig.1.), which are connected through the reduced crankshaft and presenting the elastic constantsm c_{43}, c_{32}, c_{21}, c_{1v}, dynamic absorber is attachend to the last reduced mass m_v and the only harmonic x acts upon the mechanic system applying on the reduced mass m_2.

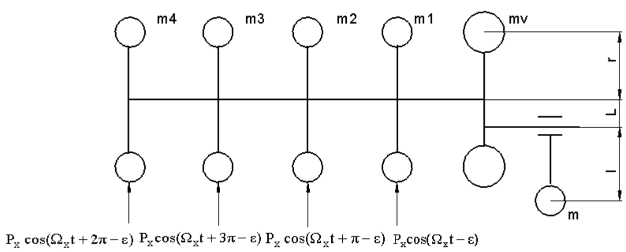

Fig.1 Mechanical system

Next we replace the dynamic absorber with an equivalent system formed of the reduced mass m_5 and the elastic shaft of a negligible mass with the elastic constant c_{v5} (Fig.2).

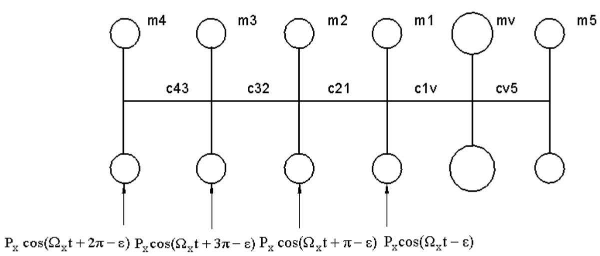

Fig.2 Equivalent mechanical system

We write the differential equations that guvern the vibratory torsion movements that the five reduced masses mechanical system with mechanical scheme presented in figure 2 [1-6]:

$$m_1\frac{d^2a_1}{dt^2}+c_{21}(a_1-a_2)+c_{1v}(a_1-a_v)=P_x\cos(\Omega_x t-\varepsilon)$$

$$m_2\frac{d^2a_2}{dt^2}+c_{21}(a_2-a_1)+c_{23}(a_2-a_3)=P_x\cos(\Omega_x t+\pi-\varepsilon)$$

$$m_3\frac{d^2a_3}{dt^2}+c_{43}(a_3-a_4)+c_{32}(a_3-a_2)=P_x\cos(\Omega_x t+3\pi-\varepsilon)\qquad(1)$$

$$m_4\frac{d^2a_4}{dt^2}+c_{43}(a_4-a_3)=P_x\cos(\Omega_x t+2\pi-\varepsilon)$$

$$m_v\frac{d^2a_v}{dt^2}+c_{1v}(a_v-a_1)+c_{v5}(a_v-a_5)=0$$

$$m_5\frac{d^2a_5}{dt^2}+c_{v5}(a_5-a_v)=0$$

The expressions for the elongations a_i (i=1~5) for the torsion vibrations executed by the six reduced masses measured on the reduction circle with a radius r, are the followings:

$a_1 = A_1\cos(\Omega_x t-\varepsilon)$

$a_2 = -A_2\cos(\Omega_x t-\varepsilon)$

$a_3 = -A_3\cos(\Omega_x t-\varepsilon)$

$a_4 = A_4\cos(\Omega_x t-\varepsilon)$

$a_v = A_v\cos(\Omega_x t-\varepsilon)$

$a_5 = A_5\cos(\Omega_x t-\varepsilon)$ (2)

The elastic constants of the segments of crankshafts between the two consecutive reduced masses and the mechanical axial moments of the reduced masses in relation to the symmetrical geometry axis of the shaft must be chosen so that the kinetic energy and the potential energy of the real vibrant system (formed by the crankshaft and its mobile gears) is equal to the kinetic energy and the potential energy of the reduced vibrant system.

In order for the mechanical system formed from the mass m_5 and the part of the shaft with the elastic constant c_{v5} to be dynamically equivalent with the dynamic absorber attached to the reduced mass m_v and thus to be able to apply the same torque as the dynamic absorber it is necessary and sufficient to:

$$m_5 = m\frac{(L+l)^2}{r^2}$$

$$c_{v5} = m\frac{(L+l)^2}{r^2}\frac{L}{l}\omega_0^2 \quad (3)$$

In addition, the harmonic x pulsation is expressed with

$$\Omega_x = x \cdot \omega_0 \quad (4)$$

where X represents the order of the harmonic, that is the number of oscillations that the harmonic executes in a time T_0, in which the shaft executes a complete rotation of 2π rad, and ω_0 is angular constant speed of rotation for the reduced crankshaft.

The amplitudes of the torsion vibrations executed by the five reduced masses A_i (i=1~5) are provided by the expressions:

$$A_i = \frac{\Delta_i}{\Delta} \quad (i=1\sim5) \quad (5)$$

The dynamic absorber is dimensioned in such a say that

$$\frac{L}{l} = x^2 \quad (6)$$

3. MATHEMATIC MODELATION

The mechanical system was established in such a way starting from the following hypotheses:

- the angular speed is constant, in reality the angular speed varies;
- one masse situated on the flywheel is assimilated for a dynamic absorber ;
- the mathematic modelation takes into account the system formed of the crankshaft and its mobile gearings (pistons, segments, vaults, rods, roller bearing), flywheel.

The corresponding mechanical system is the one composed of five reduced masses m_v, m_1, m_2, m_3, m_4, m_v which receives a dynamc absorber represented in figure 2. Mass m_1 is considered reduced mass associated with the first cylinder, the mass m_2 is the reduced mass corresponding cylinder 2, m_3 is the reduced mass corresponding cylinder 3, m_4 is the reduced mass corresponding cylinder 4, m_v is the reduced mass corresponding flywheel and m_5 is the reduced mass corresponding dynamic absorber (Fig.2).

Taking into account the mechanical system from chapter two of this paper, we can solve the differential equation system (1) using MathCad. This offers values for displacement, velocity and acceleration.

Mathematical modeling was done for the next parameters:

- n= 1650 rot/min
- the reduced masses corresponding to the crankshaft with the mobile crews and the flywheel are:

m_1= 2660,29 g $\quad$ m_2= 2654,62 g

m_3= 2661,40 g $\quad$ m_4= 2655,56 g

m_v= 5600 g $\quad$ (7)

-F_p= 9626 N $\quad$ (8)
with

$$F_p = \frac{\pi D^2}{4}(p - p_c) \quad (9)$$

where:
- D=73 mm
- p=24 daN/cm^2
- p_c= 1 daN/cm^2 $\quad$ (10)

The value of the reduced mass m_5 calculated with the relation (5) and specifying the following initial dimensions:

m=6,9 g $\quad$ L=85 mm

l=0,8 mm $\quad$ r=38,5 mm

result:

m_5= 76,42 g

In figure 3, figure 4, figure 5, figure 6, figure 7 and figure 8 are represented displacement of reduced masses m_1, m_2, m_3, m_4, m_v and m_5 within 1 second.

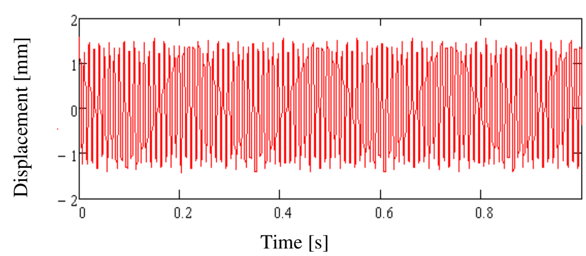

Fig. 3 Reduced mass m_1 displacement

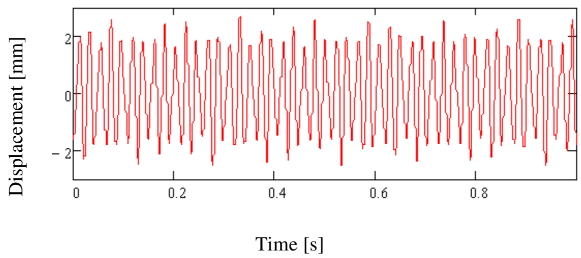

Fig. 4 Reduced mass m_2 displacement

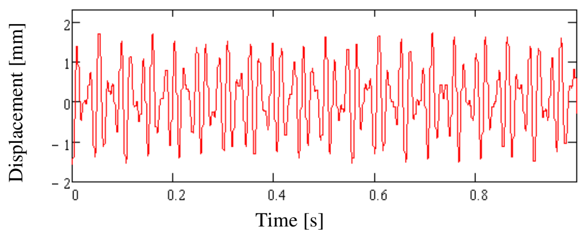

Fig. 5 Reduced mass m_3 displacement

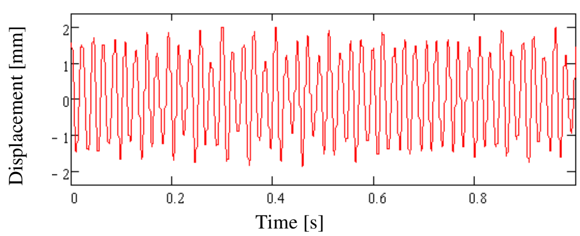

Fig. 6 Reduced mass m_4 displacement

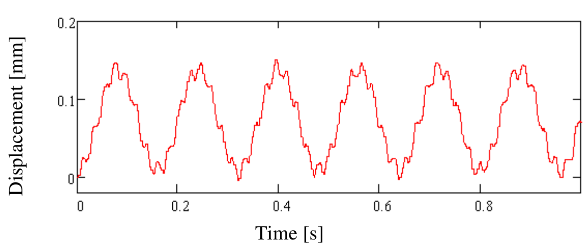

Fig. 7 Reduced mass m_5 displacement

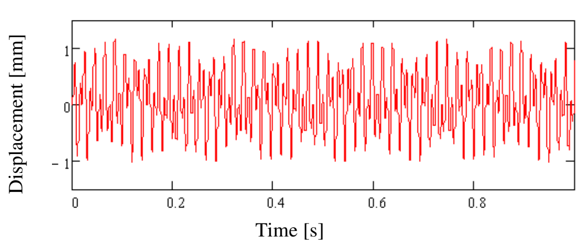

Fig. 8 Reduced mass m_v displacement

It is observed that the largest displacement are produced by reduced mass m_2.

Next we represent the velocities corresponding to the six reduced masses.

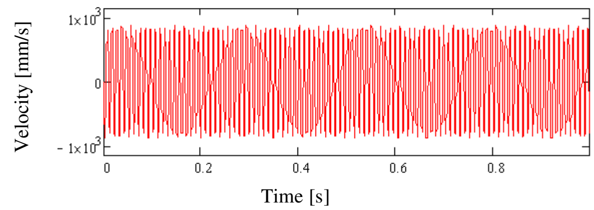

Fig. 9 Reduced mass m_1 velocity

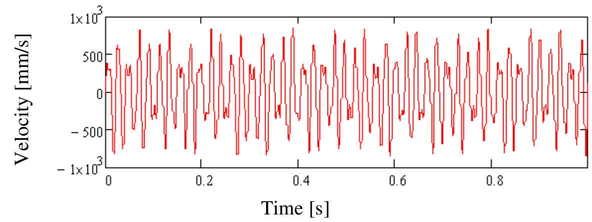

Fig. 10 Reduced mass m_2 velocity

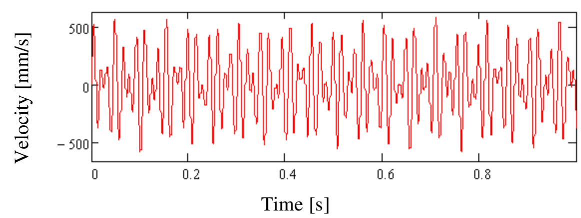

Fig. 11 Reduced mass m_3 velocity

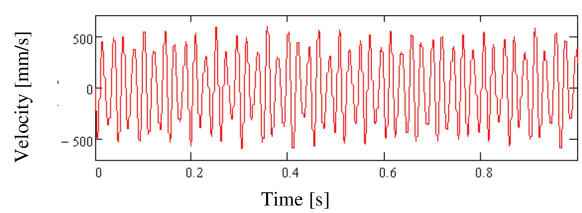

Fig. 12 Reduced mass m_4 velocity

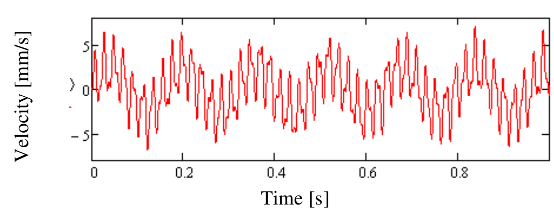

Fig. 13 Reduced mass m_5 velocity

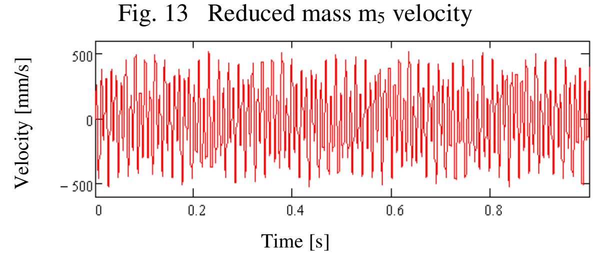

Fig. 14 Reduced mass m_v velocity

The following representations of the accelerations correspond to the six reduced masses.

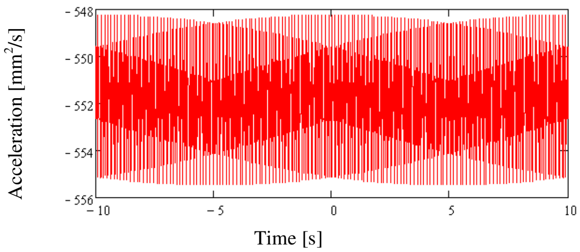

Fig. 15 Reduced mass m_1 acceleration

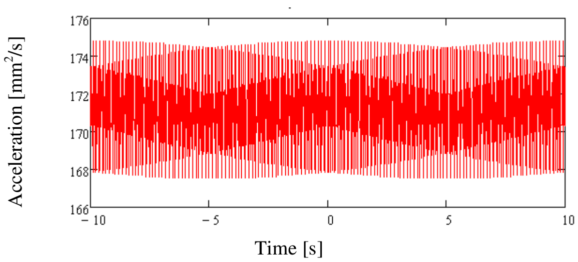

Fig. 16 Reduced mass m_2 acceleration

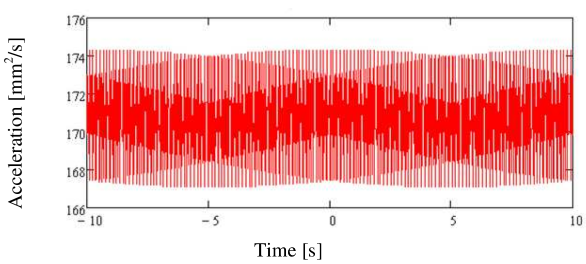

Fig. 17 Reduced mass m_3 acceleration

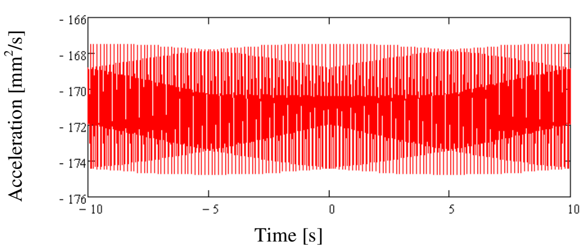

Fig. 18 Reduced mass m_4 acceleration

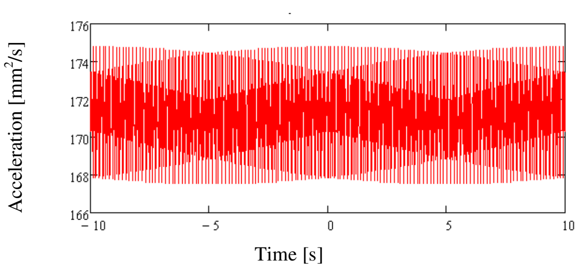

Fig. 19 Reduced mass m_v acceleration

5. CONCLUSIONS

If the dynamic absorber is built such in relation (6) it does not introduce a new resonance phenomenon and the reduced masses that come from the reduction of the crankshaft and its gearings do not perform torsion vibrations.

From a practical point of view, it is not so easy to mount a dynamic absorber on the engine flywheel that could lead to the reduction of the torsion vibrations without misbalancing the engine.

Most of the time, the dynamic absorber is superior to any isolation system because it can be easily tuned so that it works in the range of the frequencies that need to be eliminated. Using a dynamic absorber is recommended In situations where the intrinsic frequency of the primary system is close to the frequency of the disruptive forces. This is the reason that the dynamic absorber can be used to reduce unbalanced forces which appear in the functioning of various machine parts.

6. REFERENCES

[1] Ripianu A., Crăciun I., *Calculul dinamic şi de rezistenţă al arborilor drepţi şi cotiţi*, Editura Dacia, Cluj-Napoca, 1985, pag.122-126.

[2] Ripianu A., *Vibraţii mecanice, Atelierul de multiplicare al Institutului Politehnic Cluj-Napoca,1977, pag.20-22.*

[3] Bălcău, M., Ripianu, A., Case study of the mechanical system composed of four reduced masses that is subjected to a single harmonic X to wich we attached one dynamic absorber, over upon the another mass then the given two harmonic x, Acta Technica Napocensis, series Applied Mathematics and Mechanics, nr.51, vol.IV, ISSN 1221-5872, pp.99-106, 2008.

[4] Bălcău, M., Arghir, M., *The use of dynamical absorbers*, 13th International Conference „Automation in Production Planning and Manufacturing“, Zilina 2012, ISBN 978-80-89276-35-6, pag. 9-14.

[5] Bălcău, M., Opruţa, D., Pleşa, A., Bălcău, O., *Reducing the torsion vibrations with the help of a dynamic absorber*, 13th International Conference „Automation in Production Planning and Manufacturing“, Zilina 2012, ISBN 978-80-89276-35-6, pag.15-20.

[6] Bălcău, M., Pleşa, A., Opruţa, D., *Contributions to the study of diynamic absorber, particular case*, lucrare prezentată în cadrul Conferinţei ERIN 2012 25-27 aprilie 2012 organizată de Czech Technical University in Prague, Faculty of Mechanical Engineering, publicată în Acta Polytechnica Scientific Journal în august 2012, CTU Publishing House, ISSN 1805-2363.

[7] Todoruţ, A.; Cordoş, N. (2019). Evaluation of the Vehicle Sideslip Angle According to Different Road Conditions. In: Burnete N., Varga B. (eds) Proceedings of the 4th International Congress of Automotive and Transport Engineering (AMMA 2018), pp. 814-819. AMMA2018 2018. Proceedings in Automotive Engineering. Springer, Cham, Copyright information © Springer Nature Switzerland AG 2019, DOI: https://doi.org/10.1007/978-3-319-94409-8_95, First Online: 30 September 2018, Print ISBN 978-3-319-94408-1, Series Print ISSN 2524-7778, Online ISBN 978-3-319-94409-8, Series Online ISSN 2524-7786, https://link.springer.com/chapter/10.1007/978-3-319-94409-8_95, https://www.springer.com/us/book/9783319944081.

[8] Cordoş, N.; Todoruţ, A. (2019). Influences of the Suspensions Characteristics on the Vehicle Stability. In: Burnete N., Varga B. (eds) Proceedings of the 4th International Congress of Automotive and Transport Engineering (AMMA 2018), pp. 808-813. AMMA2018 2018. Proceedings in Automotive Engineering. Springer, Cham, Copyright information © Springer Nature Switzerland AG 2019, DOI: https://doi.org/10.1007/978-3-319-94409-8_94, First Online: 30 September 2018, Print ISBN 978-3-319-94408-1, Series Print ISSN 2524-7778, Online ISBN 978-3-319-94409-8, Series Online ISSN 2524-7786

[9] Iacob-Liviu SCURTU, Sanda Mariana BODEA, Ancuta Nadia JURCO, Design optimization method used in mechanical engineering. JIDEG, n. 11, p. 13-17, nov. 2016. ISSN 2344-4681.

[10] L. Scurtu, L. Banfalvi, R. Curta, A. Steopan: Using Vibrations in Smart Furniture Design, Acta Technica Napocensis-Series: Applied Mathematics, Mechanics, and Engineering, Vol.58, Issue 1, ISSN 1221-5872, Cluj-Napoca, 2015.

[11] Prodan Vasile Calin, Micle Valer, Rogozan Călin George, Szanto (Prodan) Maria – "Mathematical modeling of thermal desorption using linear regression analysis", Environmental Engineering and Management Journal, "Gheorghe Asachi" Technical University of Iasi, Romania, ISI Journal, Factor de impact 1.435, February 2013, Vol.12, No. 2, pag. 365-369, http://omicron.ch.tuiasi.ro/EEMJ/

CONSIDERAȚII TEORETICE PRIVIND ABSORBITORUL DINAMIC

Rezumat: Lucrarea de faţă studiază efectul absorbitorului dinamic asupra vibraţilor de torsiune executate de sistemul mecanic compus din masele reduse m_1, m_2, m_3, m_4, m_v supus acțiunii mai multor armonici de ordinul x. Se realizeză modelarea matemetica pornind de la ecuațiile diferențiale ce guvernează mișcarea vibratorie a sistemului mecanic propus. Sunt reprezentate grafic deplasările, vitezele și accelerațiile corespunzătoare maselor reduse.
De obicei absorbitorul dinamic este superior oricărui sistem de izolare, deoarece poat fii acordat cu uşurinţă, astfel încât să lucreze în zona frecvenţelor care trebuie eliminate. Utilizarea absorbitorului dinamic este recomandabilă când frecvenţa proprie a sistemului primar este apropiată de frecvenţa forţelor perturbatoare. absorbitorul dinamic nu majorează numărul fenomenelor de rezonanţă, oricare ar fi ordinul armonicei ce provoacă vibraţiile forţate.

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